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Finite element model updating – Case study of a rail damper

Alireza Jahan Tigh Kuchak*, Dragan Marinkovic[a] and Manfred Zehn[b]

Department of Structural Analysis, Institute of Mechanics, Technical University of Berlin, Str. d. 17. Juni 135, D - 10623 Berlin, Germany

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Abstract. In rail industry, noise reduction is a concern to decrease environmental pollution. The current study focuses on rail damper modeling and improvement of the model through validation with experimental results. Accurate modeling and simulation of rail dampers, specifically tuned rail dampers with layers interconnected by bolt joints, shall enable objective-oriented improvement of their design. In this work, to improve the damper model cone pressure theory is applied in the FE model and the sensitivity analysis is then applied to gradually improve the FE model. The improved model yields higher Modal Assurance Criterion (MAC) values and lower frequencies deviation.

Keywords: rail damper; model updating; Modal Assurance Criterion (MAC); Finite Element Modeling; eigenfrequencies; eigenvectors

1. Introduction

Damping structural vibrations is one of the most important aspects in protecting structural integrity and thus increasing safety and lifetime of structures and machines. This aspect calls, in the first place, for accurate description of structural damping properties (Mitrev and Savov 2017) and, in a further step, for developments that aim at increased structural damping (Kahya and Araz 2017, Lu *et al*. 2017, Raftoyiannis and Michaltsos 2018). Although structural acoustic emissions can even be useful for some specific purposes such as damage detection (Minak and Zucchelli 2008, Kuang *et al*. 2017, Saravanakumar *et al*. 2019), one of particular motivations for improved structural damping is actually related to attenuation of radiated noise. While some developments are based on the use of passive means such as proper choice of materials (Valvano *et al.* 2019), other developments rely on application of active means for this purpose (Gabbert *et al.* 2017, Nestorović *et al.* 2007).

Track noise belongs to the major problems in the development of railway network, particularly in urban areas. There are several sources of noise such as curve-squeal noise, bridge noise, aerodynamic noise, low frequency ground vibration noise and railway rolling noise to reduce the rolling noise. Use of rail dampers as an extra installed-mass on the rail is almost a common method in the last two decades. Many researches have utilized experimental results from field tests to find the effect of rail damper and extend its performance. However, only a sufficiently accurate damper model enables optimization of the damping effect on the rail. In particular, modeling of the tuned-damper with several rubber and steel layers proves to be rather challenging.

In 1990s the first project of noise-optimized wheels and track, called "optimized freight wheels and track (OFWHAT)" provided methods to measure the radiated noise together with methods for noise reduction in wheels and track (Thompson and Jones 2000). Wheels and track have approximately the same contribution to rolling noise (Thompson *et al*. 2007). Moreover, some inevitable effects like residual stresses during wheel manufacturing have an important influence on wheel and rail damage (Milošević *et al*. 2017), and can therewith influence the radiated noise. Zhang *et al.* (2016) have predicted the sound radiation of a railway close to the ground by boundary element method in two dimensions. This numerical prediction of the sound for both vertical and lateral motion of the rail were validated and compared experimentally. Prediction of noise has an important effect on rail damper design. Squicciarini *et al*. (2015) have studied and compared three different methods of measuring the decay rates to show the damper effect on the rail vibration in both vertical and lateral vibration. The studies focused on wheel noise and its reduction have shown the important contribution of wheels in the overall noise generated by the railway transport (Remington 1987, Thompson 1988, Thompson 1993, Fodiman *et al*. 1995, Jones *et al*. 2000). However, it is worthwhile to focus attention onto the noise radiated from the track and its component, especially rail. In this case, addition of purposefully designed dampers as extra masses on the rail helps to absorb the vibrational energy form the rail in order to reduce the vibration amplitude and therewith the noise. This approach to noise reduction at the source is economic and convenient. The achieved noise reduction can be developed and measured in both field and laboratory tests. Maes *et al*. (2003) presented a solution to reduce the noise generated by pinned-pinned frequencies on rail and showed considerable attenuation of noise at these frequencies.

*Corresponding author, Ph.D.
E-mail: alireza.jahantighkuchak@campus.tu-berlin.de
[a] Ph.D.
E-mail: dragan.marinkovic@tu-berlin.de
[b] Professor
E-mail: manfred.zehn@tu-berlin.de

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Thompson *et al*. (1996) used a theoretical model -"TWINS" to simulate and predict the rolling noise and validated these results in a field test. Liu *et al*. (2009) also proposed a theoretical model of a rail damper as a simple mass-spring model instead of an advanced beam-spring model and showed significant increase in vibration amplitude decay rate around the tuned frequency of the damper. There are different types of dampers assembled on the rail, which can reduce the vibration magnitude (Ho *et al*. 2011, Dool and Philip 2007), and in most cases dampers show effective and reasonable results (Oertli 2003). Parker and Weber (2010) showed that application of tuned-dampers with bolted layers can highly improve rail damping. Chen *et al*. (2017) have performed some experimental and dynamic characteristics optimization of tuned rail damper. They have shown a significant reduction for resonance peaks of frequency response function curve and an obvious increase of vertical transverse vibration decay rates. Assembling and manufacturing processes of dampers differ from company to company.

Setting a FE model of tuned-dampers consisting of rubber and steel layers interconnected with screws proves to be a rather challenging task. This is in particular valid when the model aims at dynamic analysis. The modal assurance criterion (MAC) values are typically used to assess the quality of an FE model by comparing the results of a numerical modal analysis with those obtained by an experiment (Ewins 2000). However, it turns out to be a fairly demanding task to obtain high MAC values for structures with bolted joints (Lifson and Smalley1989, Dunne and Heppenstall 1990, Folkman *et al.* 1995, Segalman *et al.* 2003). Bolted joints have a wide range of application in aircraft structures to assemble and fix components together and their application has a significant influence onto the static-dynamic behavior of these structures (Chen *et al.* 1995, Ireman 1998). He and Zhu (2011) developed FE models for bolted joints in L-shaped beams to capture the stiffness and mass effects of bolted joints and their influence on the global dynamic response of assembled structures. For bolted joints, they proposed a simple model which is easy to embed into the FE model of an assembled structure. They reported relatively small errors in the range of 2% for natural frequencies and MAC values above 94% in the considered frequency range. Montgomery (2002) applied three different approaches to FE modeling of bolts in bolted joints - rigid body element bolt, spider bolt and hybrid bolt were compared. A double-lap structure was modeled by Oskouei *et al.* (2009). They demonstrated crock-shaped pressure distribution at the jointed plates and longitudinal compressive stresses around the fastener hole. Marshall *et al.* (2006) applied a non-intrusive ultrasonic technique to quantify the contact pressure distribution in bolted connection. In this experimental technique, the effect of actual contact conditions can be determined. Kim *et al.* (2007) investigated several modeling techniques for bolted joints and found that the 3D solid elements and surface-to-surface contact yield the best correspondence with the test. Kapidzic *et al.* (2014) have applied a finite element modeling of fastened composite-aluminum joints in aircraft structures. They used two-node connector elements to model the fasteners, which were assigned the force-displacement characteristics determined by local models. It was shown that the fastener forces caused by temperature difference have different magnitude and have to be considered in the design of hybrid aircraft structures.

In this paper, the results of an experimental modal analysis performed on a rail damper sample from Schrey & Veit company are used to improve the FE model. FE model updating is applied as an important engineering technique (Petrovic-Kotur and Pavic, 2016). The classical theory of bolted joints is implemented in the model and the pressure cone area is parameterized while considering contact between the layers. Multiple layers with bolted joints can cause weak result in the FE model compared to experiment. However, parameterization of the clamped area around the screws opened possibilities for a significant improvement of the FE result.

2. Finite element simulation

Finite element method (FEM), as the method of choice in the field of structural analysis, is used in this work to perform numerical modal analysis of the damper. As shown in Fig.1, the damper consists of a platform and 8 additional layers – 4 steel layers and 4 rubber layers interconnected by 4 bolts and nuts. The platform is made of rubber and steel, which are vulcanized together. Commercially available FE software package Abaqus is used to develop the FE model and compute the damper's eigenfrequencies and eigenvectors.

Solid and beam elements are used in setting the FE model. The linear hexahedron element C3D8R is used to discretize the steel layers, while the quadratic hybrid hexahedron element C3D20H is used for the rubber layers. The linear beam element B31 is used to model the bolts. The element based on the hybrid formulation is used for rubber layers because this material is characterized by incompressible elastic behavior. Upon a convergence analysis, the final FE mesh used to obtain the results reported here is depicted in Fig. 1.

A special aspect of the assembled structure is the pre-stress caused by the bolt preload. Since the pre-stress affects the structural stiffness and, therewith, the eigenvalues and eigenfrequencies, the simulation is performed in two steps. The first step is performed in Abaqus as a geometrically nonlinear static general step with the loading that corresponds to the bolt preload (tightening moment). Upon this step, Abaqus saves the final configuration together with the induced stress state and this configuration is then the initial configuration of a further analysis step. In this case, the following step is the modal analysis with 'free-free' boundary conditions.

2.1 Improvement of the FE model

In the initially prepared FE model, the platform and all the layers were tied together over the whole surface for each pair of parts in contact. However, the comparison between the numerical results obtained by means of this FE model and

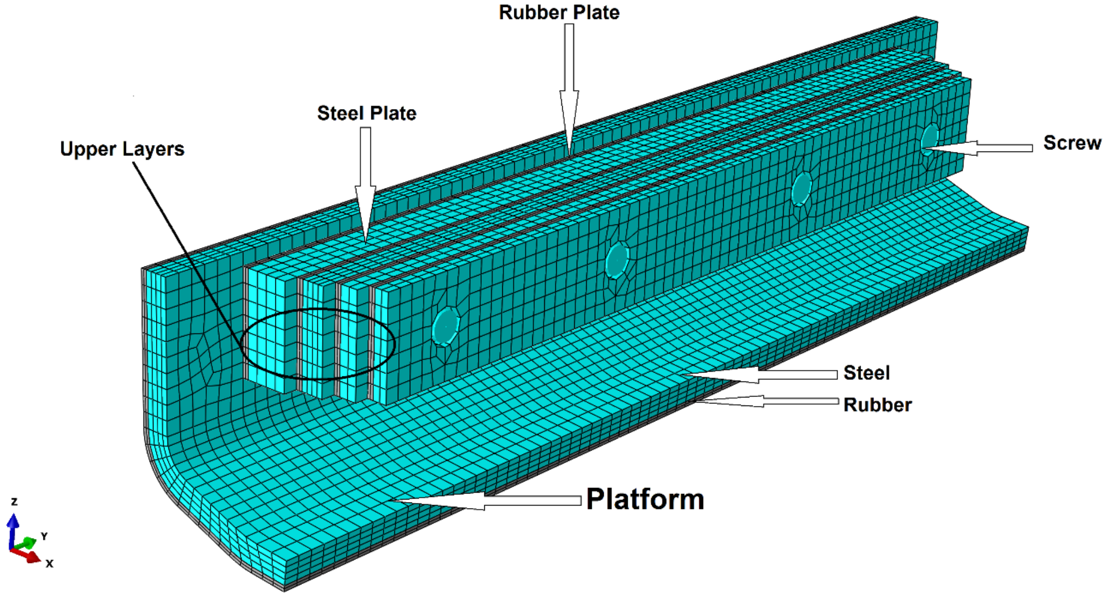


Fig. 1 FE model of rail damper with components including platform, upper layers (steel layers, rubber layers) and screws

Fig. 2 Unassembled damper to check friction between the layer

Fig. 3 Cycle test on one-meter rail and damper

the experimental results showed clearly that the FE model was deficient. The numerically determined eigenfrequencies overestimated the experimentally determined ones in the lower frequency range even by some 30%. Improvement of mesh density and changes in the applied element type for bolts resulted only in minor changes in numerical results. The parametric studies focused on mechanical properties of the rubber layers and pre-loads in bolts have neither resulted in noticeable improvements. Obviously, the obtained results suggested that the FE model was too stiff. At that point, it was clear that the FE model that implements the assumption of layers tied over the whole surface is not plausible. On the other hand, modal analysis as a linear type of analysis does not allow direct implementation of contact in the model. So, the question arose how the interaction between the layers connected by bolts is to be modeled in a plausible manner.

For this purpose, a parameterized Abaqus model was developed. The main focus of model parameterization was on bolted joints. The classical bolted joint theory determines equivalent stiffness of assembled members, which are subjected to an external tensile loading. The equivalent stiffness is derived from the stiffness of members in the clamped zone (Shigley *et al.* 2004). The distribution of stress around the bolt hole has a frustum shape, which is hollow cone in the outer layers and hollow cylinder shape at the middle layers Oskouei *et al.* (2009). Therefore, in the developed FE model, the diameter of the effective area of bolted connections is parameterized to find the value producing optimal MAC values and eigenfrequencies. For every value of the parameter, Abaqus solves the respective script, and the modal parameters are then extracted from the results (Output Data Base file) using another python script.

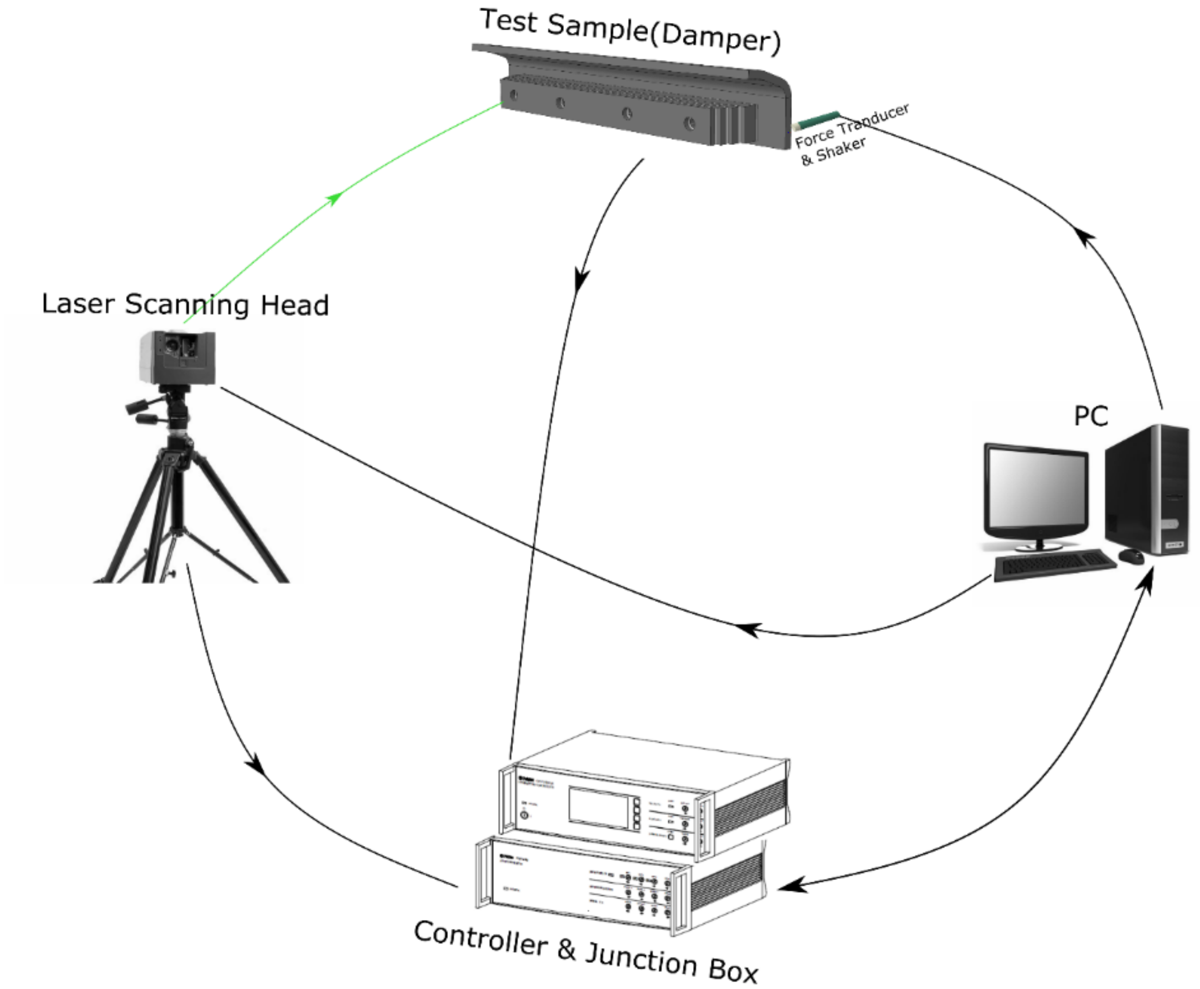


Fig. 4 Experimental equipment and schematic setup for the rail damper modal analysis

3. Experimental setup

The upper steel and rubber layers are assembled with 4 bolts. Each of them is tightened with 20 Nm tightening torque. A pre-test was first done in order to assess the importance of friction between the layers of the damper. Prior to the test, the layers of the damper were disassembled as shown in Fig. 2 Certain areas of the rubber layers, some of them in the vicinity of the bolts, while others further away from bolts, were selected to scan the surface quality by means of a microscope.

The damper was then reassembled and clamped on a one-meter rail in the laboratory. Finally, the rail with the damper was exposed to 2 million load cycles by means of a hydrodynamic press as depicted in Fig. 3. The same areas of the rubber layer surfaces were checked upon the experiment. A rather similar surface quality was observed before and after the test and no scratches could be noticed in any direction on the rubber surfaces leading to the conclusion that it was acceptable to neglect the friction between the layers in the model.

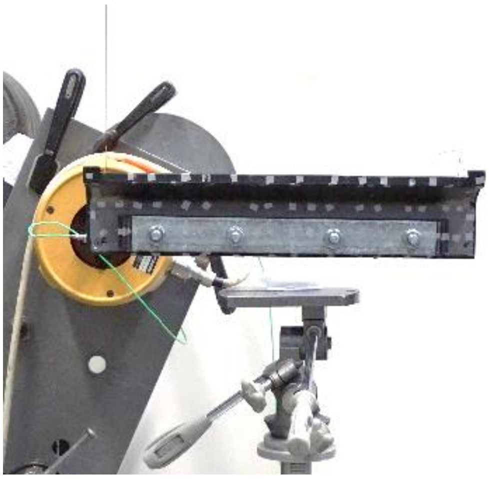
Furthermore, Polytec Scanning Vibrometer 400 (PSV) was used to perform the experimental modal analysis. The schematic modal test setup for the damper is shown in Fig. 4. The PSV measures the vibration velocities. The measured data are transferred to a computer using controllers with the velocity decoder through a junction box. The PSV software performs data acquisition and evaluates the measured data such as eigenvectors. The eigenfrequencies and damping ratios are extracted from the PSV software. Finally, software specially designed for modal analysis is used to perform curve fitting and therewith extract the modal parameters.

The scanning-software defines its own mesh of measurement points over the real structure. The laser head scanner records the velocity response of those points. High-quality reflex stickers are attached to them to facilitate the measurement. The same points are closely observed in the FE model to extract the modal parameters of the damper that are directly comparable with the experimental results. The damper is suspended using two strong but light ropes so as to have boundary conditions that are as close as possible to ‘free-free’, Fig. 5. A single-point excitation is applied in this test. The excitation is provided by shaker acting at the point located on the back side of the damper.

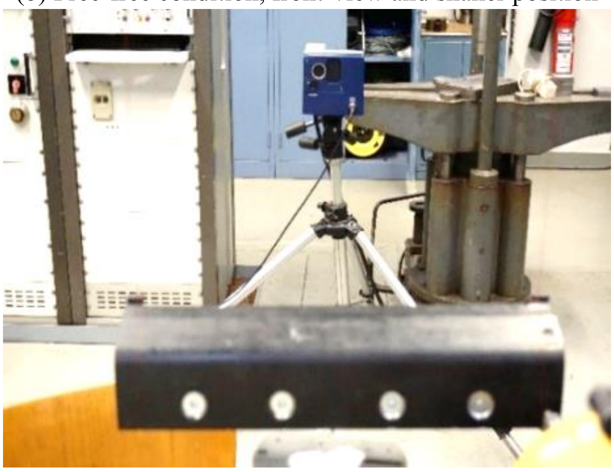
In order to isolate and better understand the issue with the initial FE model that produced low MAC values and large discrepancies between the measured and numerically determined eigenfrequencies, an additional experimental modal analysis using only the damper platform (without upper layers) was performed (see Fig. 6).

The objective of the additional tests was to separate the influence of rubber layers and bolted joints from each other. As previously said, the damper platform involves a rubber layer that is bonded to the metal part. Hence, in this case, the two layers of different materials are bonded over the whole surface. Bolts are not involved. If material nonlinearities intrinsic for rubber are the cause of the issues with the FE model, then the separate consideration of this part should reveal that.

(a) Side-view, connection force transmitter to damper



(b) Free-free condition, front-view and shaker position



(c) Back-view and laser position

Fig. 5 Experimental setup

Fig. 6 Modal test, only platform of damper

4. Result and discussion

The eigenfrequencies of the initial FE model were determined in Abaqus. They are listed in Table 1 together with the experimentally determined eigenfrequencies. At this point the correlation between the experimental and numerical modes was done visually, which is principally possible for the lower modes, but the proper way to evaluate the mode correlation will be addressed again below. The relative difference, E_r, between the numerically and experimentally determined eigenfrequencies, f_{Exp} and f_{FEM}, respectively, is determined in the straightforward manner:

$$E_r = \frac{|f_{Exp} - f_{FEM}|}{f_{Exp}} \tag{1}$$

For a proper evaluation of the degree of correlation between the experimental and numerical mode shapes, MAC analysis was employed (Allemang and Brown, 1982). For this purpose, a MATLAB script was developed. The MAC value for any pair of modes, one determined experimentally, $\emptyset_{er}$, and the other one numerically, $\emptyset_{is}$, is given in the following way:

$$MAC_{rs} = \frac{|\emptyset_{er}^T \emptyset_{is}|^2}{(\emptyset_{er}^T \emptyset_{er})(\emptyset_{is}^T \emptyset_{is})} \tag{2}$$

The ideal correlation would be represented by a MAC value equal to 1. In practice, however, some discrepancy is always to be expected which is due to a number of idealizations performed in the numerical model related to the geometry, material properties, boundary conditions, negligence of damping, etc. Nevertheless, the objective is to obtain values as close to 1 as possible across the range of the investigated mode shapes (i.e. across the investigated frequency range). A smaller difference between the numerically and experimentally determined mode shapes

Table 1 Initial FE model: eigenfrequencies and MAC values

Mode Nr.	Nature of Modes	Exp. (Hz)	FEM (Hz)	Relative Difference (%)	MAC (%)
1	1st bending	583.3	506.3	13.2	89.2
2	2nd bending	865.7	788.8	8.88	65.6
3	1st torsion	1164.6	1377.7	18.3	59.3
4	3rd bending	1618.3	1817.5	12.31	90.1
5	4th bending	2111.2	2288.8	8.41	46.3
6	5th bending	2409.8	2805	16.4	77.1
7	2nd torsion	2742.8	3280.4	19.6	67.3

Table 2 Damper platform: eigenfrequencies and MAC values

Exp. mode Nr.	FEM mode Nr.	Exp. (Hz)	FEM (Hz)	Relative Difference (%)	MAC (%)
1	1	550.4	536.89	2.4	99.59
2	2	912.2	915.72	0.3	98.91
3	4	1594.1	1556.6	2.3	99.29
4	5	2166.2	2187.6	0.9	94
5	6	2306.6	2255.9	2.1	97.39
6	7	2558.3	2546.3	0.4	90.85

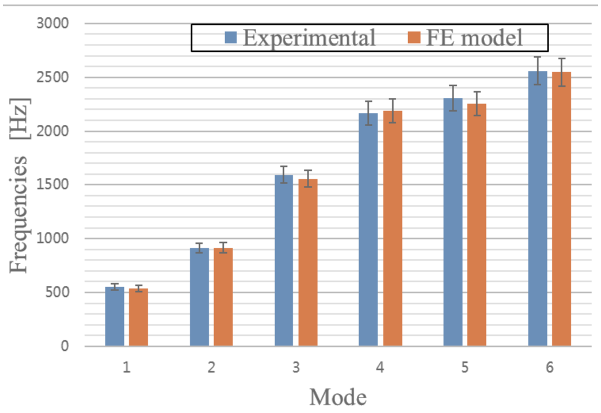


Fig. 7 Platform only: frequency-comparison for correlated mode shapes

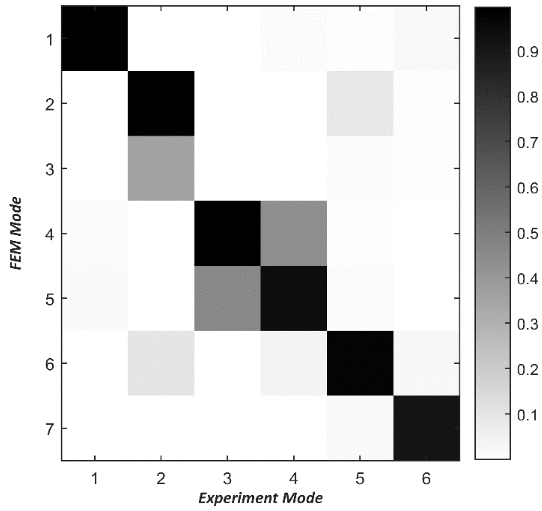


Fig. 8 Platform only: MAC results

and eigenfrequencies points to a better quality of the FE model. For the experimentally and numerically determined modes, for which initially only visual correlation was performed, Table 1 also contains the computed MAC values and the relative differences between their frequencies. Both the frequency differences and the MAC values reveal some disappointing values and highlight the drawback of purely visual correlation of the modes. Most of the MAC values are prohibitively low to consider the FE model as a valid representative of the actual structure for the purpose of engineering tasks.

As already emphasized, in order to identify the underlying problem with the developed FE model, exactly the same analysis was repeated, whereby only the damper platform (without the upper layers and bolted joints) was investigated. Table 2 presents the results that are of significantly different nature compared to those presented in Table 1. Indeed, one may see a very good correlation in both the mode shapes (MAC values) and the determined frequencies. Fig. 7 gives additionally a visual comparison of the frequencies in form of a diagram, while Fig. 8 gives the MAC matrix, i.e. the visual representation of the MAC values for all pairs of the numerically and experimentally determined mode shapes. All the results clearly confirm the very good correlation and agreement, which implies that the FE model of the damper platform alone is a good numerical representative of this part of the whole structure.

Hence, the analysis conducted so far reveals that the issue with the complete damper model is to be sought in the presence of the upper layers and the bolted-joints used to attach those layers to the damper platform. Obviously, the initial model needs careful updating, i.e. improvements.

To improve the initial FE model, the contact area between the upper layers in the cone pressure area around each bolt was parameterized. This area is of circular form around each bolt. The outer radius of this area was set as a parameter to perform a parameter study on the damper and analyse its effect onto the numerically determined eigenfrequencies and MAC values. In a python script, the radius was set as a parameter ranging from 7 mm to 27 mm and changing with a step of 0.1 mm. For each radius value, the modal analysis was performed in Abaqus and the results were used for the MAC analysis with the experimental result as a reference. The comparison between the numerically determined eigenfrequencies and experimental results was also done. Hence, the developed python script repeated the procedure of modal analysis in Abaqus and subsequent MAC analysis exactly 200 times (the range of radius between 7 mm and 27 mm, step 0.1 mm).

It would take prohibitively large space to report all the obtained results here, so the results are given here for 5 values of the considered radius values selected so as to roughly represent the development of the results for the whole range.

Table 3 Experimental and numerical eigenfrequencies for different radii in the tied zone

			FEM Results									
			R 7 (mm)		R 12 (mm)		R 17 (mm)		R 22 (mm)		R 27 (mm)	
Nr.	Mode	f_{Exp} (Hz)	f (Hz)	Δ (%)	f (Hz)	Δ (%)	f (Hz)	Δ (%)	f (Hz)	Δ (%)	f (Hz)	Δ (%)
1	1st bending	583.3	494.6	15.21	527	9.66	566.3	2.91	543.2	6.88	472.6	18.97
2	2nd bending	865.7	663.4	23.37	911.8	-5.33	857.6	0.93	885.9	2.33	937.8	8.33
3	1st torsion	1165	1127.1	3.22	1137.5	2.33	1148.9	1.35	1207.2	3.66	1308.2	12.3
4	3rd bending	1618	1289.8	20.3	1487.1	8.11	1545.5	4.5	1715.2	5.99	1765.6	9.1
5	4th bending	2111	2045.8	3.1	2090.3	0.99	2099.8	0.54	2050.4	2.88	1967.6	6.8
6	5th bending	2410	2132.7	11.5	2217.3	7.99	2364	1.9	2498	3.66	2779.2	15.3
7	2nd torsion	2743	3138.6	14.4	3177	-15.8	2916.7	6.34	3020.1	10.1	3354.2	22.3

Table 4 MAC values of damper

		MAC (%)				
Nr.	Modes	R 7 (mm)	R 12 (mm)	R 17 (mm)	R 22 (mm)	R 26 (mm)
1	1st bending	90.2	88.3	98.3	88.7	71.3
2	2nd bending	85.3	85.9	92.7	89.1	78.8
3	1st torsion	80.9	79.9	88.5	86.9	65
4	3rd bending	92.1	87.2	96	85.2	70.2
5	4th bending	76.2	65.9	89.9	74.1	58.8
6	5th bending	80.6	80.1	92.4	82.1	74
7	2nd torsion	79.8	85.3	89.8	70.7	57.4

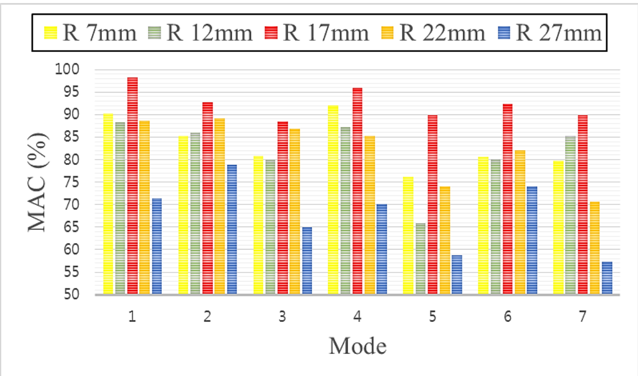


Fig. 9 MAC values of the five FE models

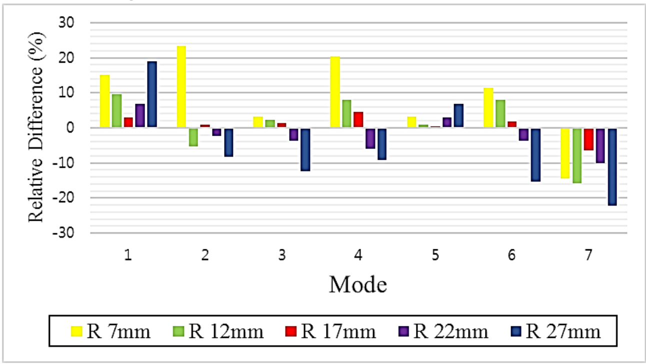


Fig. 10 Relative difference of eigenfrequencies of five FE models

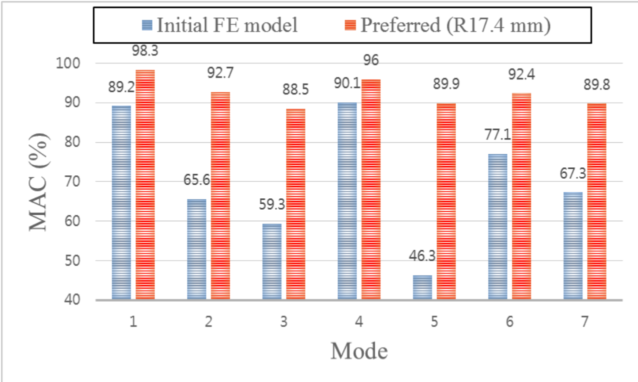


Fig. 11 Comparison of MAC values of the preferred and the initial FE model

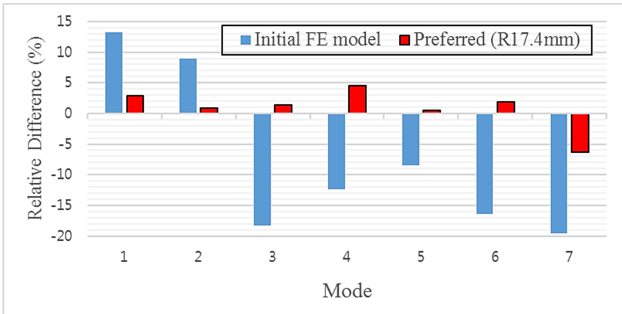


Fig. 12 Relative difference of eigenfrequencies the preferred and the initial FE model

Table 3 compares the eigenfrequencies, while Table 4 provides the MAC values. In Fig. 9 and Fig. 10, those results are represented again visually in form of diagrams - Fig. 9 shows the MAC values and Fig. 10 the differences between the numerical and experimental eigenfrequencies for the representative 5 radius values.

Obviously, among the above presented results, those for the radius of 17 mm yield the best agreement with the experimental results. As already explained, the radius was changed using the increment of 0.1 mm, so actually the best result within the whole considered set of 200 radii was obtained for the radius of 17.4 mm. Fig. 11 and Fig. 12 compare the results obtained using the initial FE model with those obtained using the model with the radius of 17.4 mm (the latter denoted as R17.4). Both the MAC values and the eigenfrequencies improved significantly.

5. Conclusions

The focus of the paper was on a tuned rail damper with bolted joint components, particularly its FE model used for modal analysis. It was first demonstrated that the initial model was not an adequate representative of the actual structure. By means of additional analyses, the culprit of the problem was identified to be in the way how the connection between the upper layers and the platform was modeled for the purpose of linear modal analysis. Upon changing the approach so that the layers were tied only in the cone pressure area around the screws, a parametric study was applied to identify the radius of the cone that produces the best results. It was shown that significantly improved results could be obtained in the whole range of he investigated eigenfrequencies.

This case study demonstrated a successful way of FE model updating. The importance of having a high quality numerical representative of the real structure comes to the fore in subsequent investigations that aim at optimized structural performance. In this specific case, further work will use the improved FE model to investigate various possibilities of increasing the noise damping performance by relatively simple structural changes. This is a work in deep progress and its results will be reported in our further publications.

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